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**PERFORMANCE OF RADIANT COOLING CEILING COMBINED WITH PERSONALIZED VENTILATION IN AN OFFICE ROOM: IDENTIFICATION OF THERMAL CONDITIONS**

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**SUMMARY**

The paper compares thermal environment conditions created by four HVAC systems: mixing ventilation, chilled ceiling combined with mixing ventilation, chilled ceiling with mixing ventilation and personalized ventilation, and chilled ceiling combined with personalized ventilation only. Measurements were performed in a test room arranged as an office with 2 workstations and 2 seating occupants resembled by thermal manikins. Heat gain of 66-72 W/m<sup>2</sup> was simulated in the room (occupants, computers, lighting, solar gain). The air temperature in the chamber was maintained at 26°C and 28°C. Personalized ventilation supplied air at non-isothermal condition with temperature of 25°C.

Results showed that the compared methods generated almost the same thermal environment in the occupied zone. However at the workstations the personalized ventilation combined with chilled ceiling provided more cooling and decreased the radiant temperature asymmetry created by the simulated warm window surface. Chilled ceiling combined with PV alone may be applied successfully in practice.

**INTRODUCTION**

Building sector uses more than 40% of the fuel and energy consumption, of which nearly half is spent on providing comfortable indoor environment. The ventilation systems currently used in buildings are mostly total volume mixing ventilation (TVMV) systems. These systems require a considerable amount of energy. They are centrally controlled and create uniform indoor environment which may not be preferred by all building occupants.

Radiant cooling systems like chilled ceiling (CC) are becoming more popular cooling solution as they remove greater part of sensible heat gains in spaces by radiation. CC has to be combined with a ventilation system to provide fresh air into the room. Most often it is mixing ventilation (MV) and rarely displacement ventilation. Another ventilation principle is personalized ventilation (PV), which, so far, has not been combined with radiant cooling. It has been documented that PV improves inhaled air quality because it delivers clean air directly to occupants' breathing zone (Melikov, 2004). The elevated air movement at the breathing zone improves perceived air quality (Melikov and Kaczmarczyk, 2012) and

occupants' thermal comfort (Kaczmarczyk et al., 2006). Previous studies performed by Kaczmarczyk et al. (2004) showed that PV decreases Sick Building Syndrome (SBS) symptoms in comparison with MV at the room air temperatures of 23°C and 26°C. Furthermore, it was shown that PV provides effective convective cooling of the body parts exposed to the air flow.

The combination of PV with CC has potential to provide more efficient cooling of occupants because of aggravation of convective and radiant cooling. PV provides increased convective cooling directly to occupants at their workstations and can be controlled according to their individual preferences (Kaczmarczyk et al., 2002). Whilst, CC creates "cool head and warm feet" environment which is more preferable by occupants than "warm head and cool feet" environment provided often by total volume systems (Imanari et al., 1999). Thus combining these two systems should be an effective way to improve the thermal comfort in rooms at higher temperature than the upper limit of 26°C as recommended in the standards (EN 15251, 2007). It was reported that increasing the maximum allowed air temperature in the room and implementing PV system may be an effective energy-saving strategy (Schiavon et al., 2010). Under certain operating conditions combination of CC with PV instead of MV is expected also to lead to reduction of the energy consumption. When the energy use is considered the variable air volume MV systems can work in a similar way as PV, but the response time of these systems is longer than the PV systems, thus the thermal comfort perceived by occupants will be reduced.

The objective of the this research was to identify and compare the environmental conditions in the room obtained by convective cooling only and convective cooling combined with radiant cooling with total volume air distribution and air distribution locally at occupied workstations. The results on the thermal environment are reported in this paper. The performance of the systems with regard to contaminants distribution is reported by Lipczynska et al. (2014).

## **METHODOLOGIES**

The thermal environment provided with total volume mixing ventilation (TVMV), chilled ceiling combined with mixing ventilation (CCMV), chilled ceiling combined with mixing ventilation and personalized ventilation (CCMV/PV), and chilled ceiling combined with personalized ventilation (CC/PV) was studied in a climate chamber arranged as an office with 2 workstations (Figure 1). At each workstation (WS1 and WS2) a personal computer was placed. Heated panels on one of the walls and on the floor simulated solar heat gains. Two thermal manikins dressed in summer clothes (total insulation with chair of 0.59 clo) were used in the experiment to simulate occupants. The body surface temperature of the manikins was controlled to be the same as the skin temperature of an average person in state of thermal comfort. The total heat loads in the room were 72 W/m<sup>2</sup> at 26°C (occupants: 66 W/manikin; computers: 65 W/computer; direct and radiant solar heat load: 654 W; lighting: 160 W) and 66 W/m<sup>2</sup> at 28°C (occupants: 53 W/manikin; computers: 65 W/computer; direct and radiant solar heat load: 553 W; lighting: 160 W).

PV used round movable panel (RMP) as an air terminal device at each workstation. The RMP was used as it has high efficiency in delivering clean air (Bolashikov et al., 2003) and allows for changing its position according to occupant preferences. PV supplied air at temperature of 25°C. Linear diffusers mounted at the centre of the ceiling were used for MV and were supplying air at difference of 10 K between supply and room air temperature. Exhaust

diffuser was located in the corner of the room at the right side from the entrance. There was no air recirculation in the systems. Chamber is equipped with 18 radiant panels which are built-in the suspended ceiling. Each panel had a dimension of 1190x590x20 mm. They totally covered 75% of floor area. The supply air flow rate for ventilation was calculated according to the recommendations in EN 15251 (2007) for category II as a sum of flow rate needed for removal of low-pollution from building materials,  $q_B$  (12 L/s), and flow rate for removal of pollution from occupants,  $q_P$  (7 L/s per person). The supply air flow for MV,  $q_{MV}$ , in CCMV case was a sum of those values,  $q_B$  and  $q_P$ , and was equal to the total supply air flow,  $q_{tot}$  (26 L/s). In cases of CCMV/PV system the MV supplied the air flow required because of building emission,  $q_B$ , and the PV supplied rate because of people occupancy,  $q_P$ . The total ventilation rate  $q_{tot}$  was a sum of flow rate supplied by the MV,  $q_{MV}$ , and by the PV,  $q_{PV}$ . The personalized air flow rate,  $q_{PV}$ , was either 7 L/s (equal to  $q_P$  required by standards) or 15 L/s per person ( $q_{P,inc}$ , increased  $q_P$  rate). In CC/PV system each RMP supplied air flow which was a sum of  $0.5q_B$  rate (required by standards for building pollution removal) and air flow rate because of occupancy:  $q_P$  or  $q_{P,inc}$ . The temperature of cooling water,  $t_w$ , circulating through CC panels depended on the heat load in a particular case studied and was set to maintain the defined air temperature constant. The total supply air flow rate in TVMV,  $q_{tot}$  equal to  $q_{MV}$ , was selected according to thermal balance to keep designed air temperature in the room.

The system configurations were compared at the following conditions: design room temperature  $t_a$  - 26 and 28 °C, air flow rate supplied by the MV,  $q_{MV}$ , and the PV,  $q_{PV1}$  at WS1 and  $q_{PV2}$  at WS2. The studied combinations are listed in Table 1.

Table 1. Designed operating parameters for cases included in evaluation

Case	Conditions						
	Design $t_a$ , °C	Total sensible heat load, W/m <sup>2</sup>	CC	MV	PV		$q_{tot}$ , L/s
			$t_w$ , °C (inlet/outlet)	$q_{MV}$ , L/s	$q_{PV1}$ , L/s	$q_{PV2}$ , L/s	
TVMV	26	72	-	82	-	-	82
CCMV	26	72	17.2/18.7	26 ( $q_B+q_P$ )	-	-	26
CCMV/PV1	28	66	20.8/21.9	12 ( $q_B$ )	7 ( $q_P$ )	7 ( $q_P$ )	26
CCMV/PV2			21.5/22.5	12 ( $q_B$ )	15 ( $q_{P,inc}$ )	15 ( $q_{P,inc}$ )	42
CCMV/PV3			21.1/22.2	12 ( $q_B$ )	15 ( $q_{P,inc}$ )	7 ( $q_P$ )	34
CC/PV1	28	66	18.7/20.3	-	13 ( $0.5q_B+q_P$ )	13 ( $0.5q_B+q_P$ )	26
CC/PV2			19.9/21.3	-	21 ( $0.5q_B+q_{P,inc}$ )	21 ( $0.5q_B+q_{P,inc}$ )	42
CC/PV3			19.4/20.9	-	21 ( $0.5q_B+q_{P,inc}$ )	13 ( $0.5q_B+q_P$ )	34

Measurements of air temperature ( $t_a$ ), globe temperature ( $t_g$ ), operative temperature ( $t_o$ ), air velocity ( $v_a$ ) and turbulence intensity ( $Tu$ ) were performed. Local discomfort due to draught and radiant temperature asymmetry was assessed. The measurements were performed in 21 locations in the room (Figure 1) at the standardized height of 0.1 m, 0.6 m, 1.1 m, and 1.7 m. At additional heights of 0.05 m, 0.3 m, 2.0 m and 2.4 m  $v_a$  and  $Tu$  were measured. The heat loss from manikins' body (whole body and body segments) was measured. For air and globe temperature measurements HOBO® U12 loggers with an accuracy of  $\pm 0.2$  K were used

(Simone et al., 2013). Low velocity multichannel thermal anemometer SENSOANEMO 5100SF with 8 omnidirectional anemometers was used for measuring air velocity with an accuracy of  $0.02 \text{ m/s} \pm 1\%$  of readings. All measured results were average of 5 min.

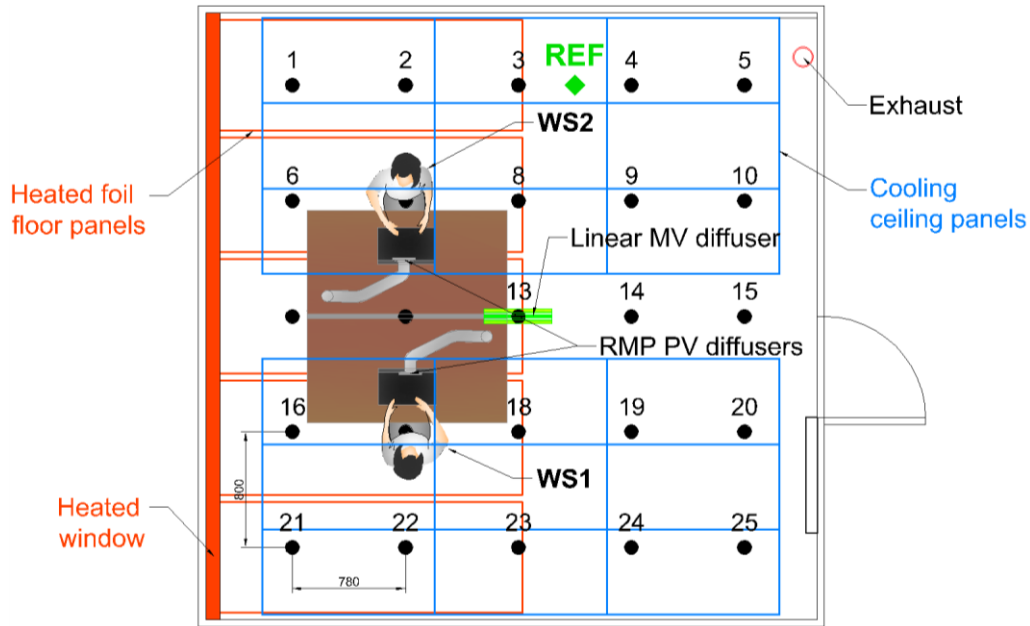


Figure 1. Lay-out of the room: plane view with measurement points and reference point (ref)

New thermal factors, TF, were defined to compare the systems effectiveness in creating uniform thermal conditions in the room (1 and 2):

$$TF_{a,j} = \bar{t}_{a,j} / t_{a,ref} \quad (1)$$

$$TF_{g,j} = \bar{t}_{g,j} / t_{g,ref} \quad (2)$$

where  $\bar{t}_{a,j}$  and  $\bar{t}_{g,j}$  are average temperature values on  $j$  height,  $t_{a,ref}$  and  $t_{g,ref}$  are temperature values in the reference point, shown in Figure 1. The thermal factor equal 1 represents uniform environment.

The equivalent temperature,  $t_{eq}$ , represents the effect of non-evaporative heat loss from the human body (ISO 14505-2, 2006). It is used to predict the average occupant's response to the thermal environment. The segmental and overall  $t_{eq}$  from thermal manikins were calculated using the following equation (3):

$$t_{eq} = t_{sk} - Q_s / h_{cal} \quad (3)$$

where  $t_{sk}$  is the manikin skin surface temperature ( $^{\circ}\text{C}$ ),  $Q_s$  is the manikin dry heat loss ( $\text{W/m}^2$ ),  $h_{cal}$  is the calibrated heat transfer coefficient ( $\text{W/m}^2\text{K}$ ).

The design air temperature, as defined in Table 1, was  $26^{\circ}\text{C}$  and  $28^{\circ}$  depending on the HVAC system. The air temperature was used as a design parameter instead of recommended in EN 15251 (2007) and EN ISO 7730 (2005) operative temperature. Decision is reasoned by heat balance calculations, where air temperature was used to define supply air temperatures for

MV and PV systems. Furthermore, traditional control systems are not equipped with operative temperature sensors but air temperature sensors. To compare the thermal performance of analysed systems, i.e. their cooling effect, the equivalent temperature was referred to the value of air temperature in the reference point (4):

$$\Delta t = t_{eq} - t_{a,ref} \quad (4)$$

## RESULTS AND DISCUSSION

The thermal environment in the room, apart from the WSs, did not show substantial differences between all analysed systems and conditions studied. Because of that only selected conditions for each system are presented in this paper (TVMV, CCMV, CCMV/PV1 and CC/PV1 – Table 1). Table 2 shows average air and globe temperature values. Effect of the radiant cooling on the globe temperature was smaller than it was expected. Differences between air temperature and globe temperature were small and in most measured points were within the accuracy of sensors. Only with TVMV system at height of 1.7 m and CCMV at 0.1 m the globe temperature was up to 0.9 K higher than the air temperature.

Table 2. Average measured values of air temperature (bold font) and globe temperature (normal font) in occupied zone

Case		TVMV	CCMV	CCMV/PV1	CC/PV1
Average temperature, °C		<b>26.1</b> / 26.4	<b>26.1</b> / 26.3	<b>28.2</b> / 28.2	<b>28.2</b> / 28.2
SD, K		<b>0.5</b> / 0.6	<b>0.4</b> / 0.5	<b>0.4</b> / 0.5	<b>0.3</b> / 0.4
Minimum temperature, °C		<b>25.3</b> / 25.4	<b>25.3</b> / 25.7	<b>27.7</b> / 27.7	<b>27.8</b> / 27.8
Maximum temperature, °C		<b>27.2</b> / 27.6	<b>27.3</b> / 27.6	<b>29.3</b> / 29.6	<b>29.2</b> / 29.5
Maximum difference, K		<b>1.9</b> / 2.2	<b>2.0</b> / 1.9	<b>1.6</b> / 1.9	<b>1.4</b> / 1.7
Average maximum horizontal difference at the same heights, K		<b>1.5</b> / 1.9	<b>1.3</b> / 1.6	<b>1.1</b> / 1.4	<b>1.0</b> / 1.3
Vertical difference 1.1 m - 0.1 m	average, K	<b>0.2</b> / 0.1	<b>0.3</b> / 0.0	<b>0.0</b> / -0.1	<b>0.0</b> / -0.1
	maximum, K	<b>0.5</b> / 0.4	<b>0.6</b> / 0.3	<b>0.2</b> / 0.3	<b>0.3</b> / 0.2
	PD, %	<b>0.5</b> / -	<b>0.5</b> / -	<b>0.4</b> / -	<b>0.4</b> / -

The vertical distribution of thermal factors  $TF_a$  and  $TF_g$ , calculated according to equations 1 and 2 at height of 0.1 m, 0.6 m, 1.1 m and 1.7 m, indicates that all systems create environment with thermal parameters close to design values. At all measured cases factors achieved values of 0.99-1.01. Vertical differences between head and ankle levels (for seated person 1.1 m and 0.1 m) were at all cases within high expectations' category A (EN ISO 7730, 2005). The maximum predicted PD caused by it was 0.5% at cases TVMV and CCMV.

The highest values of the draught rate (DR) in the occupied zone appeared in the reference cases TVMV and CCMV, where air flows supplied through MV were higher. In those cases DR was within range of 13.9-21.7%. In case CCMV/PV DR was within 5.1-10.7% and with CC/PV within 2.9-13.3%. In most cases the highest air velocity values in occupied zone were measured at height of feet (0.05-0.1 m) and were in the range of 0.20-0.33 m/s. Vertical profiles of average velocity were similar in all cases (average velocity in the range 0.10-0.16 m/s). This indicates small differences in the air flow pattern in the room between the studied systems. It is estimated that the biggest influence on the air flow pattern in the room has the convective plume from the heat sources, WSs (occupants with computers) and solar gains (warm window and heated by the solar radiation floor).

Globe temperatures measured at the locations at height of 0.6 m (the abdomen level of a seated person) were equal to operative temperatures. For all systems the impact of chilled ceiling was not big. Differences between air temperature and operative temperature at height of 0.6 m were within  $\pm 0.2$  K, which is equal to the accuracy of the sensors. Figure 2 shows operative temperature distribution in the room. It can be seen that warm surface of the window results in higher operative temperature at the part of the room near to it than in the rest of the room. Differences in the operative temperature on the two sides of the WSs were in the range 0.7-1.1 K and may be expected to cause local thermal discomfort of occupants. This non-uniformity was studied in more details with thermal manikins and is presented later in this paper.

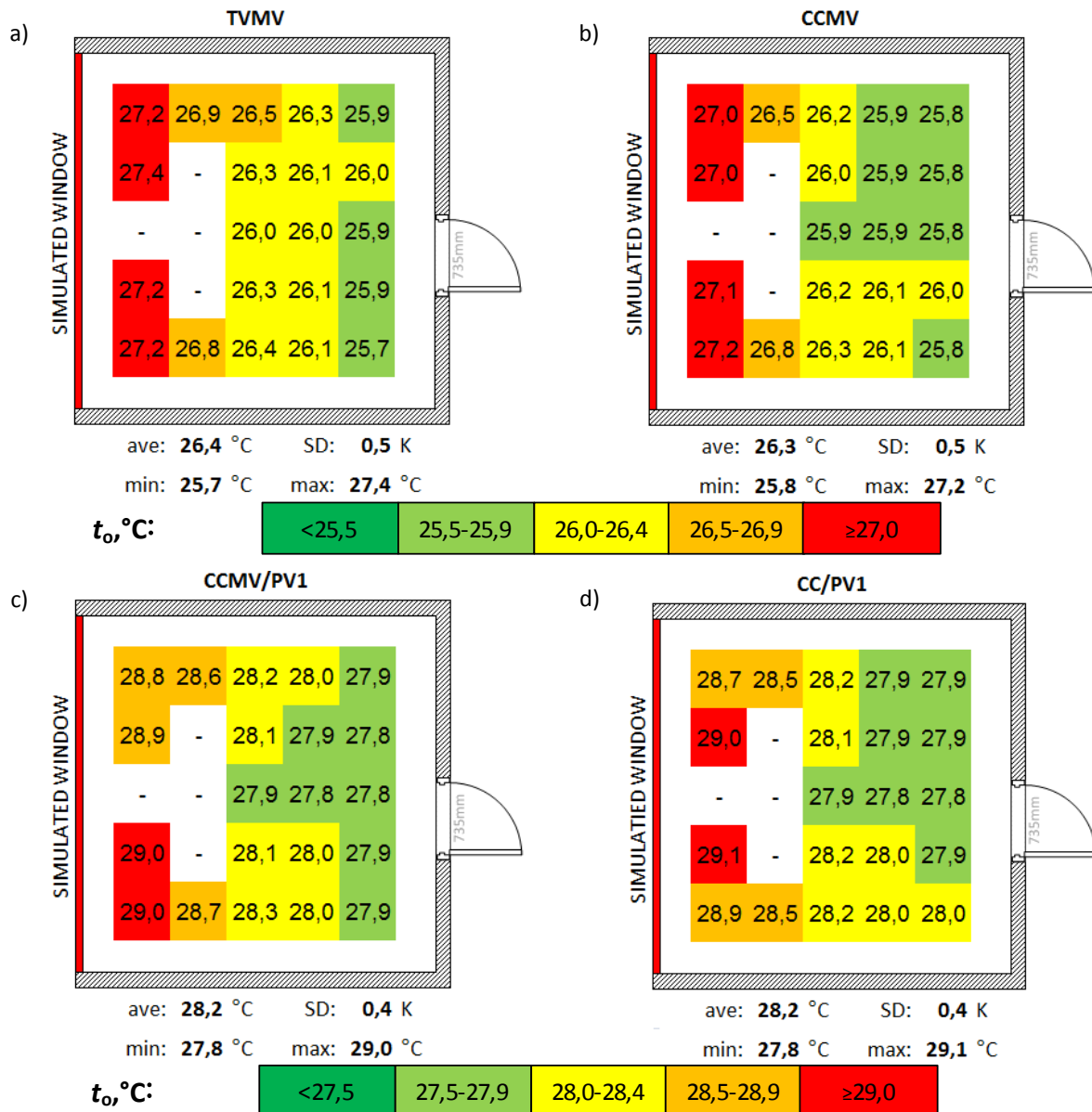


Figure 2. Plane distribution of operative temperature at 0.6 m: a, b) design  $t_a$  26°C; c, d) design  $t_a$  28°C. Color scale presents distribution in 0.5 K steps. Average (ave), minimum (min), maximum (max) and standard deviation (SD) values at this level are presented

Figure 3 shows the influence of the analysed systems on the cooling effect ( $\Delta t$ ) determined for the selected parts of human body. Results from case CCMV/PV3 were equal to the case CCMV/PV2, and from the case CC/PV3 were equal to CC/PV2. Therefore the cases CCMV/PV3 and CC/PV3 are not presented in Figure 3. It can be seen that compared to the other systems the PV cools most the upper part of the body (head, chest, neck and upper arms). Depending on the flow of supplied air through the PV the cooling effect at the face was from -2.4 K to -14.0 K at WS1 and from -5.0 K to -9.5 K at WS2. Comparing to the reference cases TVMV and CCMV the difference in the cooling effect of PV was from -0.4 K at air flow of 7 L/s to -14.0 K at 21 L/s. Use of the PV resulted in whole body cooling effect up to -0.8 K at the highest air flow from RMP of 21 L/s.

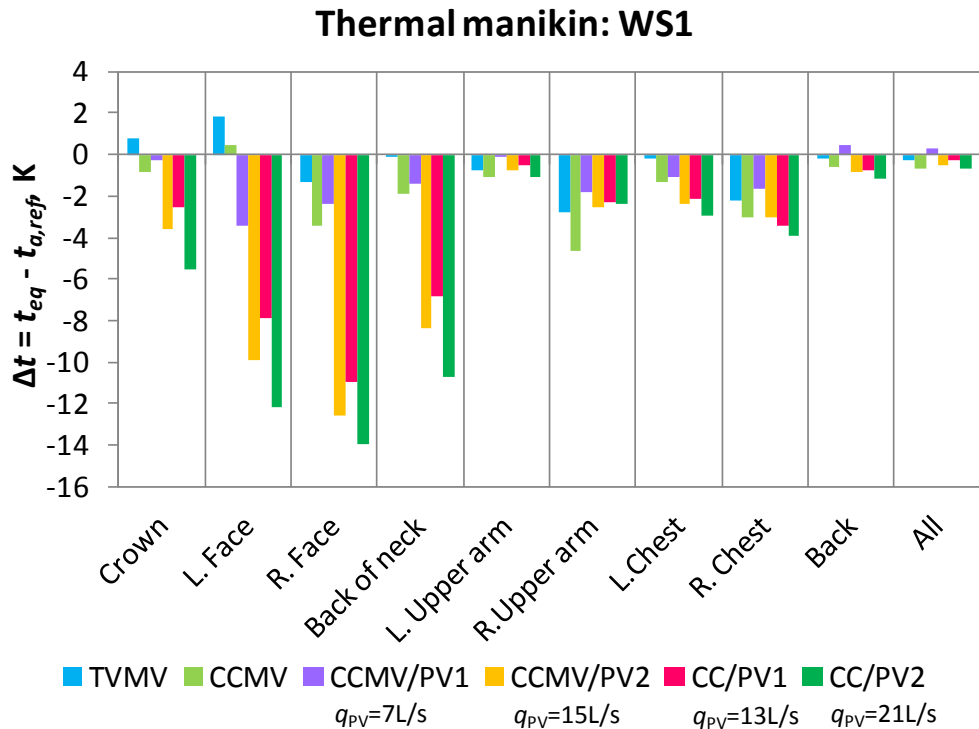


Figure 3. Cooling effect determined with thermal manikin at WS1 (results at WS2 are analogous to WS1) for different cases studied

Results obtained with both thermal manikins showed a thermal asymmetry between left and right body sides as a result of the localization of WSs next to the window. For upper arms at the reference cases TVMV and CCMV the difference in equivalent temperatures was 2.0-3.6 K at WS1 and 2.6-2.8 K at WS2. Use of the PV decreased it to values from 1.3 K to 1.9 K at both WSs depending on the air flow rates supplied through RMP.

## CONCLUSIONS

The study shows the advantages of combining personalized ventilation with cooling ceiling. Based on the obtained results following conclusions were made:

1. All systems generated similar environment in the occupied zone apart of the workstations;
2. PV combined with CC provided more cooling for the whole body and for the body segments in comparison with the remaining systems;



3. Compare to the other studied systems PV combined with CC decreased the non-uniformity in the body heat loss at the workstations caused by the simulated warm window;
4. Chilled ceiling combined with PV alone may be applied successfully in practice and will be superior to chilled ceiling combined with mixing ventilation.

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